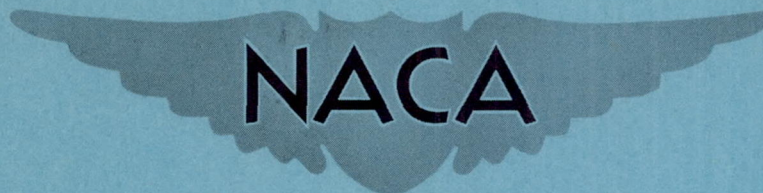


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RESEARCH MEMORANDUM

METHOD OF DETERMINING CENTRIFUGAL-FLOW-COMPRESSOR
PERFORMANCE WITH WATER INJECTION

By Joseph T. Hamrick and William L. Beede

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METHOD OF DETERMINING CENTRIFUGAL-FLOW-COMPRESSOR PERFORMANCE
WITH WATER INJECTION

By Joseph T. Hamrick and William L. Beede

SUMMARY

A method is presented for computing the isentropic and actual enthalpy change between the inlet and the outlet of a centrifugal-flow compressor when water injection is used. The change in state of the water in the working fluid complicates the use of specific heats in accounting for property changes in the fluid. Consequently, the method of calculation is based on fluid property values given in steam and air tables. Functional charts are included to show the effect of water-air ratio on isentropic outlet temperature, isentropic enthalpy rise, and adiabatic efficiency with various pressure ratios for a fixed set of inlet conditions.

For a given isentropic enthalpy change or adiabatic efficiency, the rate of increase in the compressor pressure ratio decreased as the water-air ratio was increased. For compressor-inlet conditions of pressure, 14 inches of mercury absolute; temperature, 77° F; and specific humidity, 0, water-air ratios greater than 0.05 were relatively ineffective for pressure ratios of less than 8. On the basis of the large difference in flow conditions between wet and dry compression, the design of a compressor that would have good efficiency for both wet and dry compression might be difficult.

INTRODUCTION

Water-methanol injection at the compressor inlet is a standard thrust-augmentation method for turbojet engines. The large increase in thrust obtainable with this method has resulted in investigations of the performance effects of water injection on the engine components to determine possible sources of losses. Difficulty in evaluating the properties of the vapor-air mixture complicates the analysis of the turbojet cycle and especially the computation of compressor efficiency.

A method of computing the efficiency of a centrifugal compressor with water injection at the inlet was derived at the NACA Lewis

laboratory and is presented herein. Inasmuch as the isentropic-adiabatic compression process is used in computing efficiency with dry compression, the same process is used in this method. Conditions of the water-air mixture at the inlet are arbitrarily set in order to simplify the method. Although this method can be applied to mixtures that include injection liquids for which extensive thermodynamic data are available, water alone is used in the numerical example in order to simplify computation. Because the water contained in the air changes to vapor during the compression process, the latent heat of vaporization must be accurately evaluated. Another complication is the variation in the specific heat of the vapor, which precludes the use of constant specific heats in calculating efficiency. The use of variable specific heats would be cumbersome and impractical. For computing changes in enthalpy trial-and-error methods based on the steam and air tables of references 1 and 2 were used to eliminate the use of variable-specific-heat formulas.

The method outlined herein was used to compute theoretical values needed for evaluating performance. Functional charts are given for isentropic outlet temperature, isentropic enthalpy change, and adiabatic efficiency for a range of pressure ratios and fixed inlet conditions. Water-air ratio is plotted as a parameter in all the charts.

SYMBOLS

The following symbols are used in the calculations and the figures:

c_p	specific heat at constant pressure, (Btu/(lb)(°F))
h	enthalpy, (Btu/lb)
N	ratio of actual pressure to base pressure
P	total pressure, (lb/sq in.) or (in. Hg absolute)
q	specific humidity, (lb water/lb air)
R	universal gas constant for air, 53.35 (ft-lb/(lb)(°F))
s	entropy, (Btu/(lb)(°F))
T	temperature, (459.7 + t) (°R)
t	temperature, (°F)

- v specific volume, (cu ft/lb)
- W air weight, (lb)
- w/a water-air ratio due to water injection
- η_{ad} adiabatic efficiency

$$\varphi = \int_{T_0}^T \frac{c_p}{T} dT, \text{ (Btu/(lb)(}^\circ\text{F))}$$

The following subscripts refer to states of working fluid through the compressor:

- 0 NACA standard sea-level conditions
- 1 compressor inlet before injection of water
- 2 compressor outlet
- a constituent of mixture based on 1 pound of dry air
- d dry air
- f saturated liquid
- g saturated vapor
- m mixture
- s superheated vapor
- t stagnation

ANALYSIS

In order to enable comparisons of compressor performance with wet- and dry-compression, in the following analysis adiabatic efficiency is based on the isentropic-adiabatic compression process used for dry compression. The adiabatic efficiency of a compressor is defined as the ratio of the isentropic (reversible adiabatic) compression work to the actual compression work. Although the actual work can be determined by either thermodynamic or mechanical methods, the isentropic work must be computed by use of thermodynamics.

If conditions at the outlet are such that the entropy at the outlet equals that at the inlet, the second law of thermodynamics precludes the possibility of any change in entropy during the process; therefore only the end points of the process are considered. Evaluation of the entropy of the mixture at the inlet is difficult because an increase in entropy due to adiabatic saturation of the air at the inlet cannot be avoided. Only if the air is saturated before the water is injected and is at the same temperature as the injected water can the process conceivably be ideally reversible. For a fixed set of inlet conditions and given compression ratio, the pressure at the outlet is fixed. The temperature at the outlet will therefore determine not only the state of the injected liquid but also the entropy at the outlet. A temperature that will give an entropy at the outlet equal to that at the inlet must therefore be found by trial and error. When this temperature is found, the enthalpy of the outlet mixture can be determined.

From the Gibbs-Dalton law it may be deduced that the entropy and the enthalpy of a mixture of gases and vapors can be regarded as the sum of the individual entropies and enthalpies of the constituents when each occupies the same space as the mixture at the temperature of the mixture. This law is not extended to liquid particles suspended in a gas. For the method presented herein, however, the liquid-water particles are assumed to be homogeneously distributed throughout the mixture but to exert no pressure. The entropy and the enthalpy for the particles are calculated in the same manner as for gases.

In computing the entropy of the mixture at the inlet, no attempt is made to estimate the amount of water evaporated into the air on injection. Instead, the entropy of the mixture is based on conditions existing before any degree of vaporization of injected water occurs. The assumption of no vaporization at the compressor inlet simplifies the calculation procedure. Data based on no saturation and complete saturation of the inlet air indicate only a small difference in enthalpy change; however, the condition of no saturation results in a more conservative value of efficiency. The temperature of the water particles is assumed equal to that of the water before injection in order to approximate more closely the value of the inlet entropy of the mixture. All values of entropy are taken from the steam and air tables of references 1 and 2, respectively. The entropy of each constituent in the mixture is based on 1 pound of dry air, with the result that the sum of all the entropies gives the entropy of the mixture per pound of dry air. The enthalpy of the mixture at the inlet is determined by the same procedure.

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In order to compute the outlet temperature for which the entropy at the outlet equals that at the inlet, an outlet temperature is first assumed. With the assumed temperature and the water-air ratio at the outlet, the entropy is computed in the same manner as for the inlet. When the isentropic outlet temperature is found, either of two sets of conditions may prevail at the outlet depending on the quantity of water injected: The air may be saturated at the outlet with or without water droplets being present, or the vapor may be in the superheated condition, in which case no droplets will be present. In the case of saturation with or without water droplets, the entropy can be easily found because these properties are the same as those for saturation temperature. With the superheated vapor, the exact temperature at which saturation occurs must be determined. The pressure corresponding to the temperature of saturation is the same that the vapor exerts at any higher temperature, the pressure of the mixture remaining constant. When this pressure is determined, the entropy of the vapor can be obtained from the superheated-vapor tables of reference 1. The enthalpy of the mixture at the outlet for either the isentropic or the actual case is determined by the same procedure.

If the mixture at the outlet contains droplets in a mixture of superheated vapor and air, the actual outlet temperature cannot be used. The compressor work, or actual enthalpy rise, must then be determined by mechanical means. An estimate of the quantity of droplets present can be made by the method illustrated in the Numerical Example if the dry-bulb temperature at the outlet is known. The humidity due to the superheated vapor at the outlet can be approximated by trial and error by finding a vapor-air ratio at the outlet that gives an enthalpy change equal to the power input. The procedure is the same as that for calculating the actual outlet temperature except that the vapor-air ratio must be varied for succeeding computations. Any water not in the superheat state is considered to be in droplet form.

NUMERICAL EXAMPLE

A numerical example is presented to show the method of computing isentropic enthalpy rise, actual enthalpy rise, and adiabatic efficiency of a compressor for a given set of conditions. In each case where a temperature is assumed for trial-and-error solution, only the calculations resulting in the correct temperature are shown. The following conditions from actual experimental data prevail:

Water injected, lb/sec	2.175
Air weight flow, lb/sec	43.97
Water temperature, °F	55
Air temperature, °F	77.4
Inlet pressure, lb/sq in.	6.870
in. Hg abs.	14.00
Outlet pressure, lb/sq in.	19.998
in. Hg abs.	40.73
Outlet temperature, °F	264
Specific humidity ^a , lb water/lb air	0.01025
Water-vapor pressure (humidity ^a), lb/sq in.	0.1110

^aThe specific humidity and the pressure of water vapor at the inlet are calculated from an assumed wet-bulb temperature by means of the empirical equations of reference 3.

I. Entropy of Mixture at Inlet

(1) Water-air ratio due to injection. -

Pounds water contained in inlet air

$$(0.01025)(43.97) = 0.4507 \text{ (lb water/sec)}$$

Pounds dry air supplied

$$43.97 - 0.4507 = 43.52 \text{ (lb/sec)}$$

Water-air ratio due to injection

$$\begin{aligned} w/a &= 2.175/43.52 \\ &= 0.04997 \end{aligned}$$

(2) Entropy due to injected liquid. -

Entropy of saturated liquid at inlet

$$s_{f,1} \text{ at } 55^\circ \text{ F} = 0.0459 \text{ (Btu/(lb)(}^\circ\text{F))}$$

Entropy of saturated liquid per pound of dry air at inlet

$$\begin{aligned} s_{f,a,1} &= s_{f,1} w/a \\ &= (0.0459)(0.04997) \\ &= 0.002294 \text{ (Btu/(lb)(}^\circ\text{F))} \end{aligned}$$

(3) Entropy due to superheated vapor in incoming air. - Although the Mollier diagram for steam does not include values of entropy for a pressure of 0.1110 pound per square inch, the low-pressure region near saturation indicates an entropy increase of 0.018 Btu per pound per $^{\circ}\text{F}$ for each temperature rise of 20°F along an isobaric line. By using the entropy at saturation for a pressure of 0.1110 pound per square inch and correcting for temperature difference from saturation temperature to 77.4°F , an entropy value $s_{s,1}$ of 2.204 Btu per pound per $^{\circ}\text{F}$ was obtained.

Entropy of superheated vapor per pound of dry air at inlet

$$\begin{aligned} s_{s,a,1} &= s_{s,1} \quad q_1 \\ &= (2.204)(0.01025) \\ &= 0.02259 \text{ (Btu/(lb)(}^{\circ}\text{F))} \end{aligned}$$

(4) Entropy due to dry air. -

Pressure of dry air

$$\begin{aligned} P_{d,1} &= P_1 - P_{s,1} \\ &= 6.870 - 0.1110 \\ &= 6.759 \text{ (lb/sq in.)} \end{aligned}$$

$$\frac{1}{N} = \frac{P_0}{P_{d,1}}$$

$$= 14.70/6.759$$

$$= 2.175$$

$$R \log_e \frac{1}{N} = - R \log_e N = - 0.05326$$

where

$$R = 53.35 \text{ (ft-lb/(lb)(}^{\circ}\text{F))}$$

The temperature function at the inlet $\varphi_{t,1}$ is 0.07067, from table 1 of reference 2.

Entropy of dry air at inlet

$$\begin{aligned}s_{d,1} &= \phi_{t,1} - R \log_e N \\&= 0.07067 + 0.05326 \\&= 0.1239 \text{ (Btu/(lb)(}^\circ\text{F))}\end{aligned}$$

(5) Entropy of mixture at inlet. - Because ratios of the quantities remain unchanged, the use of tables having different temperature bases (references 1 and 2) is valid. The entropy of mixture per pound of dry air equals the entropy of saturated liquid per pound of dry air plus the entropy of superheated vapor per pound of dry air plus the entropy of dry air.

$$\begin{aligned}s_{m,a,1} &= s_{f,a,1} + s_{s,a,1} + s_{d,1} \\&= 0.002294 + 0.02259 + 0.1239 \\&= 0.1488 \text{ (Btu/(lb)(}^\circ\text{F))}\end{aligned}$$

II. Isentropic Outlet Temperature

(1) Outlet temperature. - An outlet temperature $t_{t,2}$ of 108.1°F is assumed for computing an entropy at the outlet equal to that at the inlet by trial-and-error solution.

(2) Specific humidity at outlet. -

$$\begin{aligned}P_{g,2} \text{ at } t_{t,2} \text{ of } 108.1^\circ\text{F} \\&= 1.206 \text{ (lb/sq in.) (from table 1, reference 1)}\end{aligned}$$

$$P_2 = 19.998 \text{ (lb/sq in.) (original conditions)}$$

$$\begin{aligned}P_{d,2} &= P_2 - P_{g,2} \\&= 19.998 - 1.206 \\&= 18.79 \text{ (lb/sq in.)}\end{aligned}$$

$$\begin{aligned}v_{g,2} \text{ at } t_{t,2} \text{ of } 108.1^\circ\text{F} \\&= 279.5 \text{ (cu ft/lb) (from table 1, reference 1)}\end{aligned}$$

$$T_{t,2} = 567.8 \text{ }^{\circ}\text{R (given)}$$

$$R = 53.35 \text{ (ft-lb/(lb)(}^{\circ}\text{F))}$$

$$\begin{aligned} W &= \frac{P_{d,2} v_{g,2}}{R(T_{t,2})} \\ &= \frac{(144)(18.79)(279.5)}{(53.35)(567.8)} \\ &= 24.97 \text{ (lb)} \end{aligned}$$

where

W weight of air that would occupy same volume as $v_{g,2}$

Specific humidity at outlet

$$\begin{aligned} q_2 &= \frac{1}{W} \\ &= \frac{1}{24.97} \\ &= 0.04005 \text{ (lb water/lb air)} \end{aligned}$$

(3) Quantity of liquid water present at outlet. - The total water-air ratio equals the water-air ratio due to water injected plus the water-air ratio due to humidity at inlet.

$$\begin{aligned} \text{total water-air ratio} &= w/a + q_1 \\ &= 0.04997 + 0.01025 \\ &= 0.06022 \end{aligned}$$

$$\frac{\text{pound liquid water}}{\text{pound dry air}} \text{ at outlet} = \text{total water-air ratio} -$$

specific humidity at outlet

$$\begin{aligned} &= 0.06022 - 0.04005 \\ &= 0.02017 \end{aligned}$$

(4) Entropy due to liquid water. -

Entropy of saturated liquid at outlet

$$s_{f,2} \text{ at } t_{t,2} \text{ of } 108.1^{\circ}\text{F}$$

$$= 0.1438 \text{ (Btu/(lb)(}^{\circ}\text{F)) (table 1, reference 1)}$$

Entropy of saturated liquid per pound of dry air at outlet

$$s_{f,a,2} = s_{f,2} \left(\frac{\text{lb liquid water}}{\text{lb dry air}} \right)$$

$$= (0.1438)(0.02017)$$

$$= 0.002900 \text{ (Btu/(lb)(}^{\circ}\text{F))}$$

(5) Entropy due to saturated vapor. -

Entropy of saturated vapor at outlet

$$s_{g,2} \text{ at } t_{t,2} \text{ of } 108.1^{\circ}\text{F}$$

$$= 1.962 \text{ (Btu/(lb)(}^{\circ}\text{F)) (table 1, reference 1)}$$

Entropy of saturated vapor per pound of dry air at outlet

$$s_{g,a,2} = s_{g,2} q_2$$

$$= (1.962)(0.04005)$$

$$= 0.07859 \text{ (Btu/(lb)(}^{\circ}\text{F))}$$

$$N = \frac{P_{d,2}}{P_0} = \frac{18.79}{14.70} = 1.278$$

$$R \log_e N = 0.01681$$

Temperature function at outlet

$$\varphi_{t,2} = 0.08403 \text{ (Btu/(lb)(}^{\circ}\text{F)) (from table 1, reference 2)}$$

Entropy of dry air at outlet

$$\begin{aligned}s_{d,2} &= \phi_{t,2} - R \log_e N \\&= 0.08403 - 0.01681 \\&= 0.06722 \text{ (Btu/(lb)(}^{\circ}\text{F))}\end{aligned}$$

(6) Entropy of mixture at outlet. - The entropy of mixture per pound of dry air at the outlet equals the entropy of saturated liquid per pound of dry air plus the entropy of saturated vapor per pound of dry air plus the entropy of dry air.

$$\begin{aligned}s_{m,a,2} &= s_{f,a,2} + s_{g,a,2} + s_{d,2} \\&= 0.002900 + 0.07859 + 0.06722 \\&= 0.1487 \text{ (Btu/(lb)(}^{\circ}\text{F))}\end{aligned}$$

When $s_{m,a,1}$ and $s_{m,a,2}$ are equal, the assumed temperature is that for isentropic compression. If they are unequal, a new temperature $t_{t,2}$ must be selected until equality is attained. The entropy value as computed is sufficiently close to that at the inlet to assume an isentropic process.

III. Enthalpy Change Through Compressor

(1) Enthalpy at inlet. -

Enthalpy of saturated liquid at inlet

$$h_{f,1} \text{ at } t_{f,1} \text{ of } 55^{\circ} \text{ F} = 23.07 \text{ (Btu/lb)}$$

Enthalpy of saturated liquid per pound of dry air at inlet

$$\begin{aligned}h_{f,a,1} &= h_{f,1} w/a \\&= (23.07)(0.04997) \\&= 1.153 \text{ (Btu/lb)}\end{aligned}$$

Enthalpy of saturated vapor at inlet

$$h_{g,1} = 1096 \text{ (Btu/lb) at } p_{s,1} \text{ of } 0.1110 \text{ and } t_{t,1} \text{ of } 77.4^{\circ} \text{ F}$$

Pressure effect on enthalpy is negligible in this region of pressure.

Enthalpy of saturated vapor per pound of dry air at inlet

$$\begin{aligned} h_{g,a,1} &= h_{g,1} q_1 \\ &= (1096)(0.01025) \\ &= 11.23 \text{ (Btu/lb)} \end{aligned}$$

Enthalpy of dry air at inlet

$$h_{d,1} \text{ at } T_{t,1} \text{ of } 537.1^\circ \text{ R} = 32.87 \text{ (Btu/lb)}$$

The enthalpy of mixture per pound of dry air at the inlet equals the enthalpy of saturated liquid per pound of dry air plus the enthalpy of saturated vapor per pound of dry air plus the enthalpy of dry air

$$\begin{aligned} h_{m,a,1} &= h_{f,a,1} + h_{g,a,1} + h_{d,1} \\ &= 1.153 + 11.23 + 32.87 \\ &= 45.25 \text{ (Btu/lb)} \end{aligned}$$

(2) Enthalpy at outlet. -

Enthalpy of saturated liquid at outlet

$$h_{f,2} \text{ at } t_{t,2} \text{ of } 108.1^\circ \text{ F} = 76.05 \text{ (Btu/lb)}$$

Enthalpy of saturated liquid per pound of dry air at outlet

$$\begin{aligned} h_{f,a,2} &= h_{f,2} \left(\frac{\text{lb liquid water}}{\text{lb dry air}} \right) \\ &= (76.05)(0.02017) \\ &= 1.534 \text{ (Btu/lb)} \end{aligned}$$

Enthalpy of saturated vapor at outlet

$$h_{g,2} \text{ at } t_{t,2} \text{ of } 108.1^\circ \text{ F} = 1108.6 \text{ (Btu/lb)}$$

Enthalpy of saturated vapor per pound of dry air at outlet

$$\begin{aligned} h_{g,a,2} &= h_{g,2} q_2 \\ &= (1108.6)(0.04005) \\ &= 44.40 \text{ (Btu/lb)} \end{aligned}$$

Enthalpy of dry air at outlet

$$h_{d,2} \text{ at } T_{t,2} = 40.25 \text{ (Btu/lb)}$$

The enthalpy of mixture per pound of dry air at the outlet equals the enthalpy of saturated liquid per pound of dry air plus the enthalpy of saturated vapor per pound of dry air plus the enthalpy of dry air.

$$\begin{aligned} h_{m,a,2} &= h_{f,a,2} + h_{g,a,2} + h_{d,2} \\ &= 1.534 + 44.40 + 40.25 \\ &= 86.18 \text{ (Btu/lb)} \end{aligned}$$

(3) Enthalpy change from inlet to outlet. -

$$\begin{aligned} \Delta h_{m,a} &= h_{m,a,2} - h_{m,a,1} \\ &= 86.18 - 45.25 \\ &= 40.93 \text{ (Btu/lb)} \end{aligned}$$

If the vapor at the outlet is in the superheat region, the entropy and the enthalpy are computed for the assumed temperature in the same manner as the enthalpy or the actual work in section IV.

IV. Actual Work Determined from Total Temperature at Outlet

$$\text{Temperature at outlet} = 264^{\circ} \text{ F}$$

For this temperature, the vapor contained in the air is in the superheated condition. The pressure exerted by the vapor in the mixture will therefore be equal to that exerted at saturation. The following method is used to determine the temperature at which saturation occurs. By assuming values of saturation temperature t

and using the corresponding values of specific volume v_g found in table 1 of reference 1, a value of specific humidity equal to the total water-air ratio at the outlet is obtained.

From section II, item (3) the total water-air ratio is 0.06022. Assume the temperature t of saturation to be 121.3°F . The pressure $P_{g,2}$ corresponding to 121.3°F is 1.755 pounds per square inch in the steam tables (table 1, reference 1).

(1) Pressure of superheated vapor. -

$$\begin{aligned} P_{d,2} &= P_2 - P_{g,2} \\ &= 19.998 - 1.755 \\ &= 18.24 \text{ (lb/sq in.)} \end{aligned}$$

The volume occupied by 1 pound of vapor at saturation v_g for 121.3°F is 196.5 cubic feet per pound from table 1, reference 1.

The weight of air that would occupy the same volume as v_g at $P_{d,2}$ is

$$\begin{aligned} W &= \frac{P_{d,2} v_g}{R(t + 459.7)} \\ &= \frac{(144)(18.24)(196.5)}{(53.35)(581.0)} \\ &= 16.65 \text{ (lb)} \end{aligned}$$

Water-air ratio

$$\frac{1}{W} = \frac{1}{16.65} = 0.0601$$

Inasmuch as the actual $q_{m,2}$ is 0.06022 pound water per pound air, the assumed temperature is acceptable; the pressure exerted by the vapor in the superheat region $P_{s,2}$ is therefore 1.755 pounds per square inch.

(2) Enthalpy of mixture. -

Enthalpy of vapor h_g at 264°F = 1179 (Btu/lb) (reference 1, fig. 10)

Enthalpy of vapor per pound of dry air at outlet

$$h_{s,a,2} = (0.0601)(1179) = 70.86 \text{ (Btu/lb)}$$

Enthalpy of dry air

$$h_{d,2} \text{ at } 264^{\circ} \text{ F} = 77.85 \text{ (Btu/lb)}$$

Enthalpy of mixture per pound of dry air at outlet

$$\begin{aligned} h_{m,a,2} &= h_{s,a,2} + h_{d,2} \\ &= 70.86 + 77.85 \\ &= 148.7 \text{ (Btu/lb)} \end{aligned}$$

$$h_{m,a,1} = 45.25 \text{ (Btu/lb) from section III}$$

(3) Enthalpy change equivalent to actual work. -

$$\begin{aligned} \Delta h_{m,a} &= h_{m,a,2} - h_{m,a,1} \\ &= 148.7 - 45.25 \\ &= 103.5 \text{ (Btu/lb)} \end{aligned}$$

V. Adiabatic Efficiency

$$\begin{aligned} \eta_{ad} &= \frac{\text{isentropic } \Delta h_{m,a}}{\text{actual } \Delta h_{m,a}} \\ &= \frac{40.93}{103.5} = 0.395 \end{aligned}$$

APPLICATION OF METHODS

Isentropic outlet temperature, isentropic enthalpy change, and adiabatic efficiency computed for various pressure ratios are shown in figures 1, 2, and 3, respectively. Water-air ratios from 0 to 0.06 are also given. Inlet conditions of pressure, 14 inches of mercury absolute; temperature, 77° F; and specific humidity, 0 are

used in each case. For any change in inlet conditions, additional curves must be constructed. The slip factor in figure 3 is defined as the ratio of the rotational velocity of the fluid particle at the outlet to tangential velocity of the impeller tip.

In figure 1, the curves for all water-air ratios closely follow the same path until the vapor in the mixture passes into the superheat state, at which point the curve breaks sharply upward. As the water-air ratio increases, the breakaway occurs at increasingly higher pressure ratios. This effect is reflected in figure 2, which shows that, for any given isentropic enthalpy change, the change in pressure ratio increases at a decreasing rate as water-air ratio is increased. For a given adiabatic efficiency in figure 3, the rate of increase in pressure ratio decreases as water-air ratio is increased. A water-air ratio greater than 0.05 is relatively ineffective for pressure ratios of less than 8 for these compressor-inlet conditions. In view of the large difference in flow conditions between dry and wet compression, the design of a compressor that will have good efficiencies for both wet and dry compression may be difficult.

For the inlet conditions used herein, figures 1 and 2 can be used as a source of data in computing efficiencies of actual compressors and in making analyses of the turbojet cycle. For the compressor used in a cycle analysis with a given pressure ratio and an assumed efficiency, the isentropic enthalpy change can be taken from figure 2 and the actual enthalpy change computed. The actual outlet temperature that will give the actual enthalpy change can be found by trial-and-error methods based on use of steam and air tables.

SUMMARY OF RESULTS

From the method developed for determining centrifugal-flow-compressor performance with water injection, it has been shown that for any given isentropic enthalpy change or adiabatic efficiency, the rate of increase in compressor pressure ratio decreased as the water-air ratio was increased. For compressor-inlet conditions of pressure, 14 inches of mercury absolute; temperature, 77° F; and specific humidity, 0, water-air ratios greater than 0.05 were relatively ineffective for pressure ratios of less than 8.

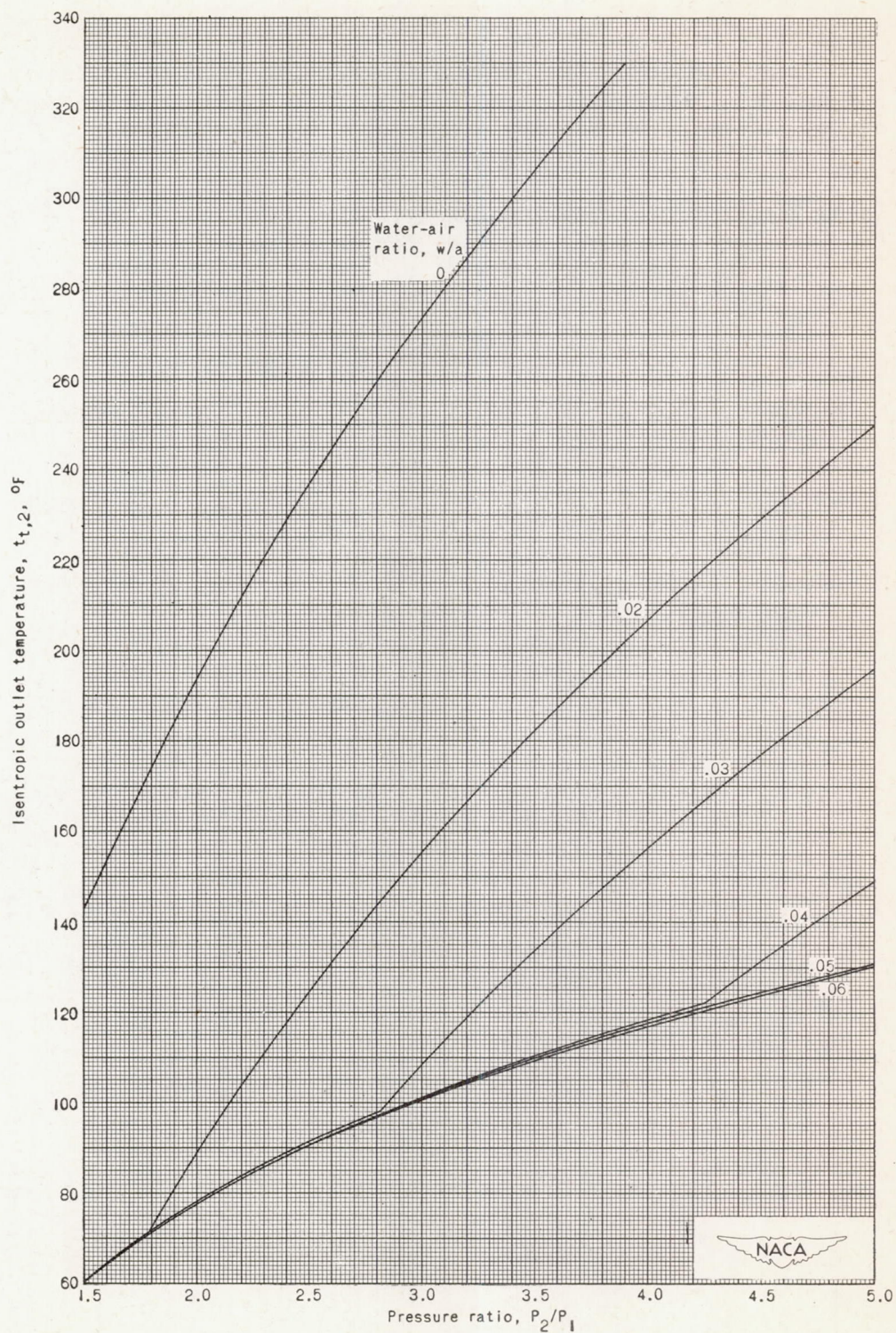
CONCLUSION

On the basis of the large difference in flow conditions between wet and dry compression, the design of a compressor with good efficiency for both wet and dry compression may be difficult.

Lewis Flight Propulsion Laboratory,
National Advisory Committee for Aeronautics,
Cleveland, Ohio.

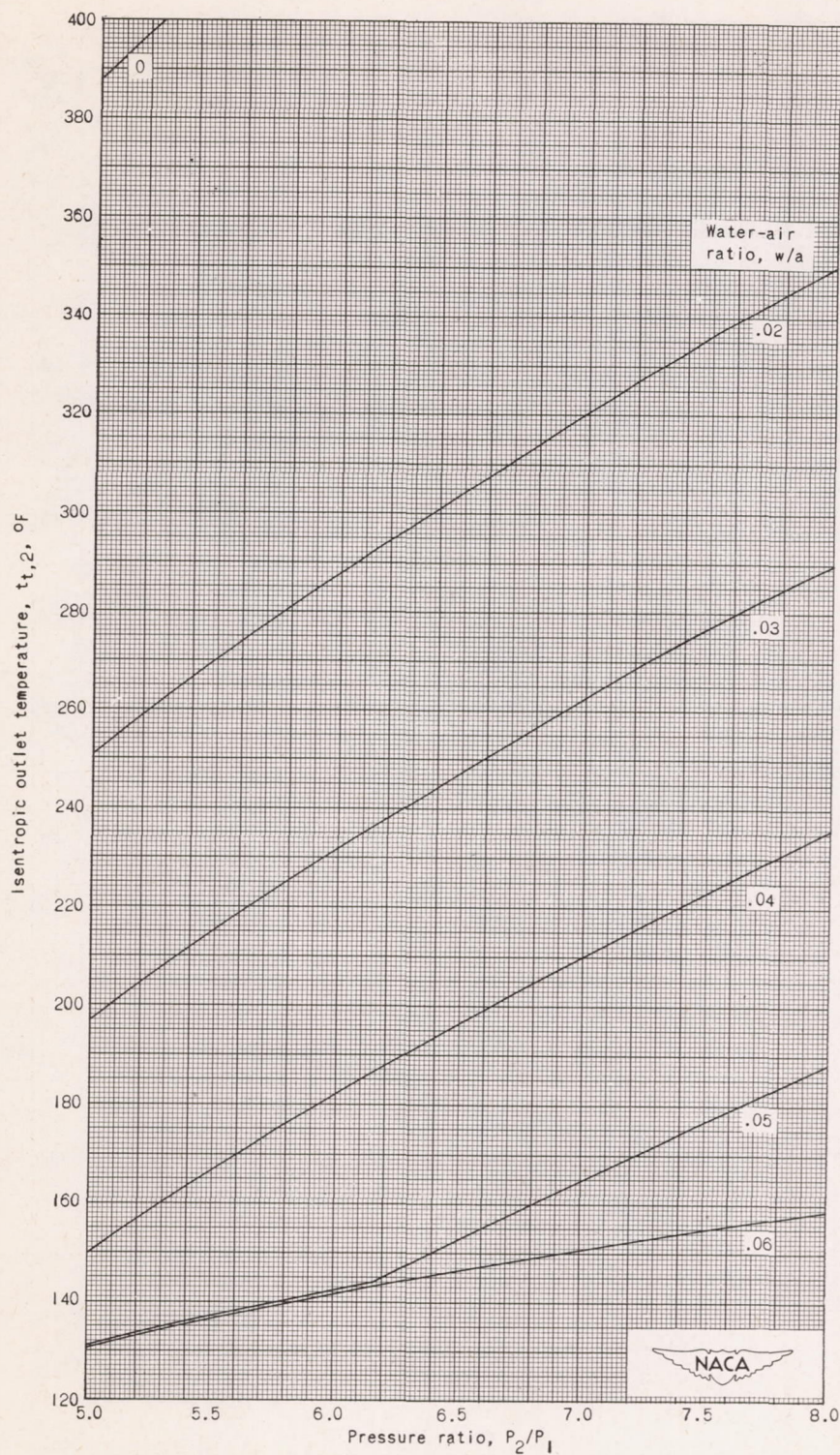
REFERENCES

1. Keenan, Joseph H., and Keyes, Frederick G.: Thermodynamic Properties of Steam. John Wiley & Sons, Inc., 1936.
2. Keenan, Joseph H., and Kaye, Joseph: Thermodynamic Properties of Air. John Wiley & Sons, Inc., 1945.
3. Anon.: The Refrigerating Data Book. Am. Soc. Refrigerating Eng. (New York), 5th ed., 1942, p. 428.



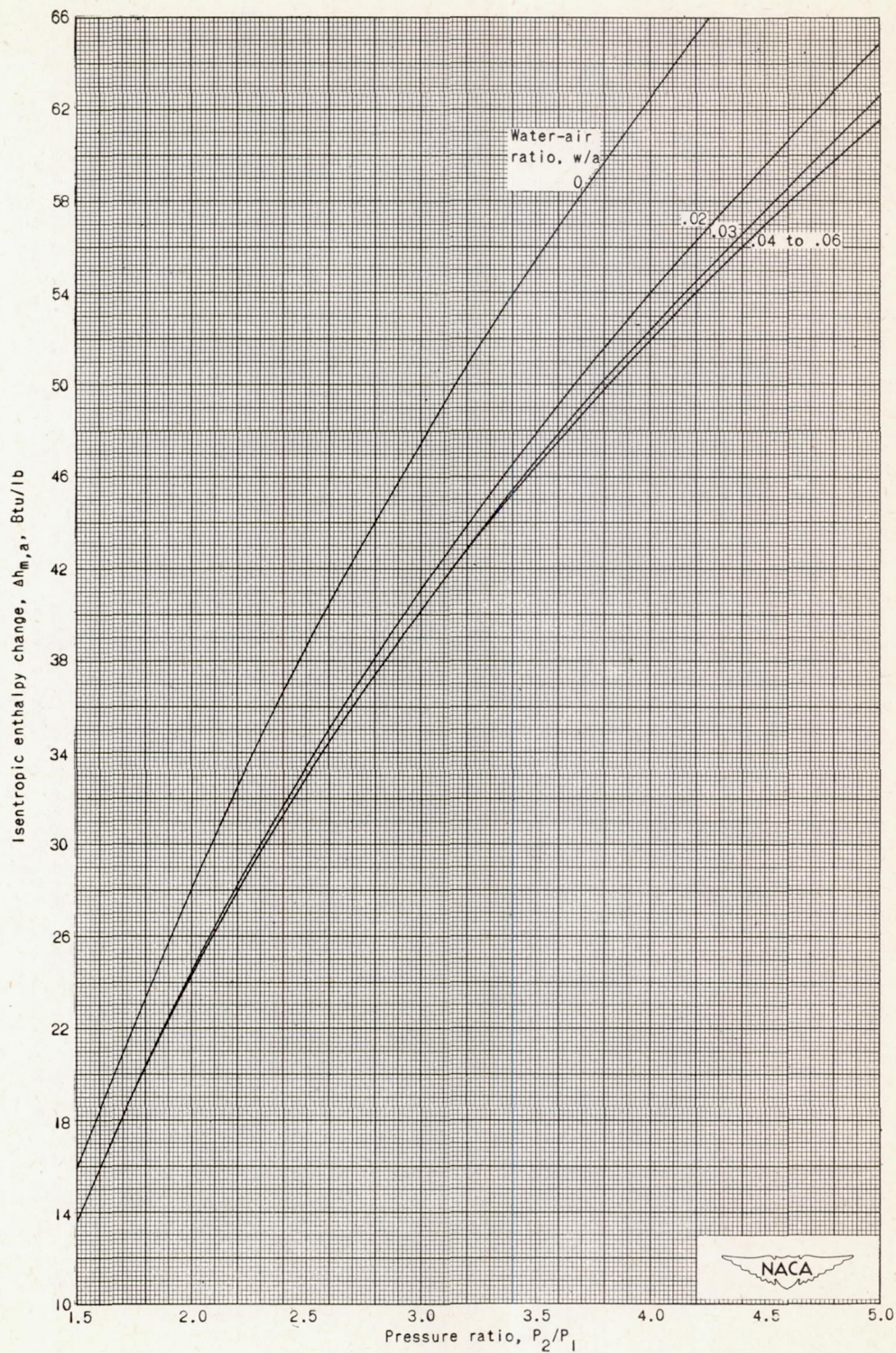
(a) Pressure ratio, 1.5 to 5.0.

Figure 1. - Variation of outlet temperature with pressure ratio for isentropic compression. Compressor-inlet conditions: pressure, 14 inches mercury absolute; temperature, 77° F; specific humidity, 0.



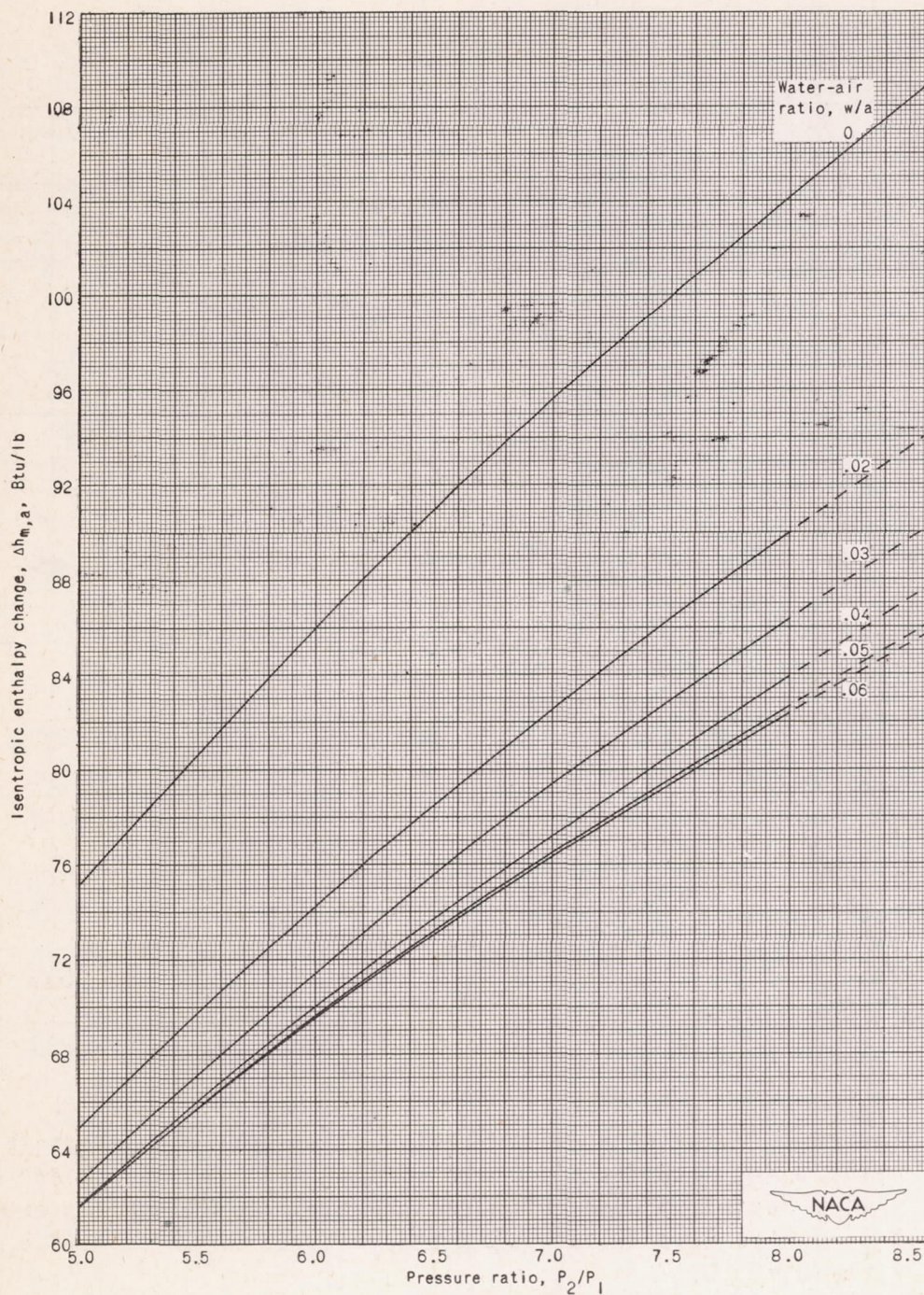
(b) Pressure ratio, 5.0 to 8.0.

Figure 1. - Concluded. Variation of outlet temperature with pressure ratio for isentropic compression. Compressor-inlet conditions: pressure, 14 inches mercury absolute; temperature, 77° F; specific humidity, 0.



(a) Pressure ratio, 1.5 to 5.0.

Figure 2. - Variation of enthalpy change with pressure ratio for isentropic compression. Compressor-inlet conditions: pressure, 14 inches mercury absolute; temperature, 77° F; specific humidity, 0.



(b) Pressure ratio, 5.0 to 8.5.

Figure 2. - Concluded. Variation of enthalpy change with pressure ratio for isentropic compression. Compressor-inlet conditions: pressure, 14 inches mercury absolute; temperature, 77° F; specific humidity, 0.

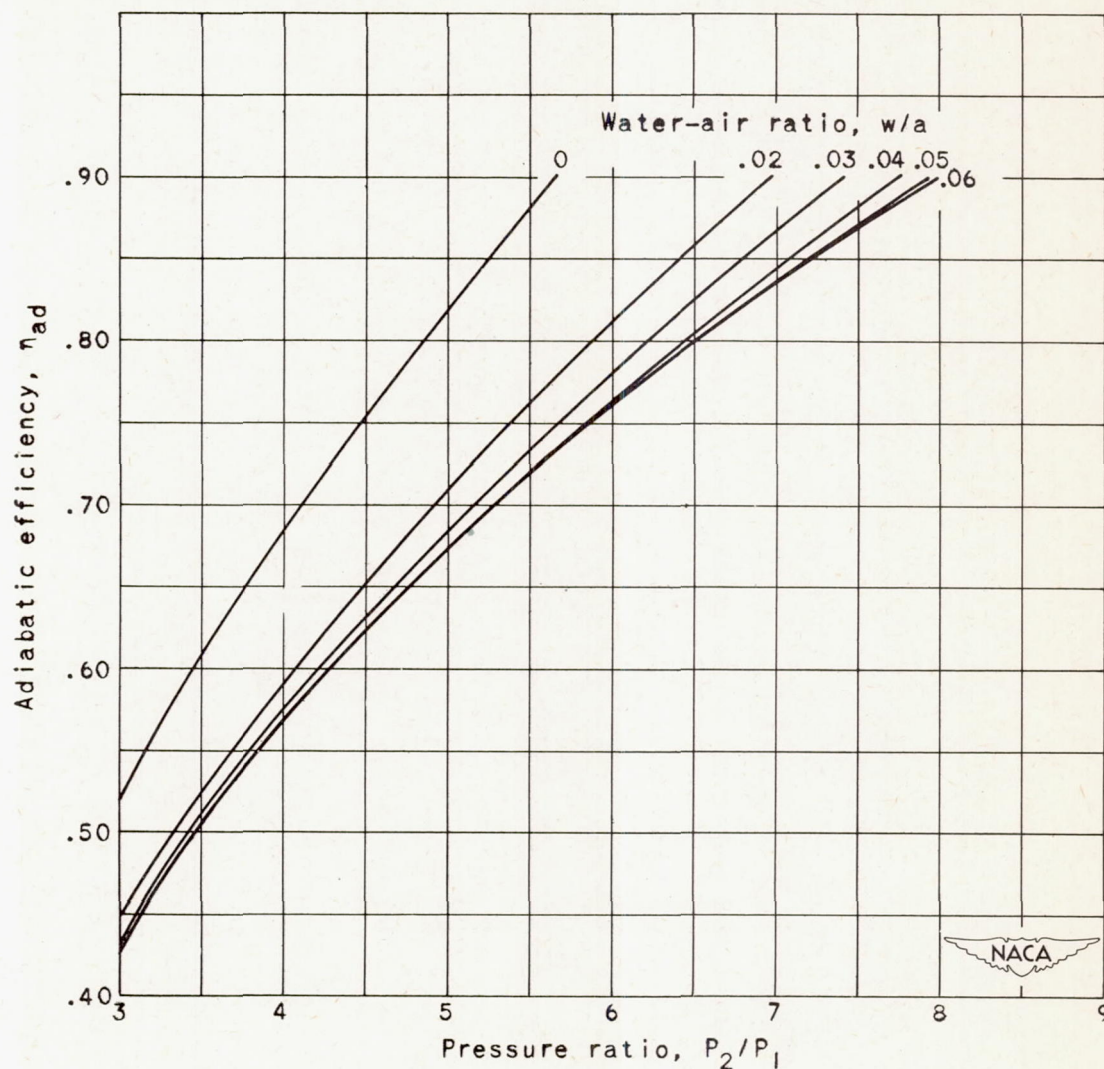


Figure 3. - Variation of adiabatic efficiency with pressure ratio for impeller tip speed of 1560 feet per second and slip factor of 0.94. Compressor-inlet conditions: pressure, 14 inches mercury absolute; temperature, 77° F; specific humidity, 0.